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Computer Simulation of Lubrication Conditions of Trunk Pistons in Refrigeration Compressors

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COMPUTER SIMULATION OF LUBRICATION CONDITIONS OF TRUNK PISTONS IN REFRIGERATION COMPRESSORS

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INTRODUCTION

Concerning the optimization of reciprocating piston engine the problems of friction lubrication and wear especially in the system of piston, piston ring, liner are of particular interest. Besides of the valves this assembly is one of the most important parts of the reciprocating refrigeration compressor. It has to fulfil its tasks to do the compression work on the refrigerant with as low frictional and gas losses and oil carryover as possible.

With regard to the reciprocating internal combustion engines a great number of experimental and theoretical investigations have been done in the area of friction and wear. These investigations show that piston, piston ring and liner are a system of a sliding bearing with mainly hydrodynamic lubrication. Only near the dead centres of the piston stroke under certain conditions nonfluid friction can occur.

Because of the principal similarity of the piston assembly of combustion engines and compressors similar mechanisms of lubrication friction and wear can be supposed. Owing to the more abundant lubrication and the lower level of temperatures and pressures in the cylinder of the refrigeration compressor the system piston, piston ring and liner can be considered more reasonable as a hydrodynamic bearing system. Additionally the constant running conditions of refrigeration compressors with exception of the start and stop of the engine are more favorable for a hydrodynamic lubrication compared with the internal combustion engines.

On the other side lubrication failure with regard to compressors occurs sometimes under special running conditions. For the purpose of the essential oil return from the refrigeration circuit to the lubricated compressor which has always small oillosses, going with the compressed refrigerant through the circuit, a low oil viscosity at low evaporator temperatures has to be demanded. Because of this essential low oil viscosity for the oil return, a much lower viscosity at running conditions in the compressor is the consequence. This low oil viscosity sometimes causes a breakdown of the lubrication of the crankshaft bearings and the bearing system piston-cylinder, especially in proximity of the application limits given by pressure ratio and discharge gas temperature.

This failure of lubrication results in a great wear on the thrust side of the piston and piston ring scuffing. Especially in using refrigeration compressors for heatpump application, because of the more severe running conditions, a breakdown of the lubrication has often occured.

Until today for compressors the design of the sliding bearing system piston-liner is mainly of an empirical nature. Therefore, a lot of experiences and experimental tests are necessary in order to lay out the shape, material and mechanical properties of piston and piston rings for optimal fulfilling the task of transfer of force and for sealing against the working fluid and the lubrication oil under all running conditions. Economical using of material, reliability and a maximum of efficiency should be generally the aim in compressor construction for reasons of economics. In the opinion of Qvale and Soedel e.a. [1] the problems of piston and piston ring friction are an important area of still open problems in mathematical modelling and simulation of refrigerating compressors, on which work should be done. Also Krug, Najork and Schulz [2] besides the design of crankshaft bearings, propose also an inclusion of the system piston cylinder in a wanted optimization of machine elements by means of a complex modelling system for compressor simulation.

For the calculation of the lubrication conditions between piston and liner a simulation program for the working cycle of a compressor is necessary. Such simulation models for this purpose have already been established [3], using the real state equations for refrigerants. Besides of metallurgic and chemical problems, the tribology of piston, piston ring and liner especially needs the theoretical calculation of the lubrication conditions using the theory of hydrodynamic lubrication on which already has been reported [4,5,7] In this paper a further extension of this hydrodynamic model of calculation shall be discussed. The purpose of this model is to calculate the lubrication conditions in the area piston, piston ring, cylinder using the digital simulation of the working cycle of refrigeration compressors.

CALCULATION OF LUBRICATION CONDITIONS

For the calculation of the lubrication conditions the knowledge of the geometry of the clearance between both sliding parts is important.

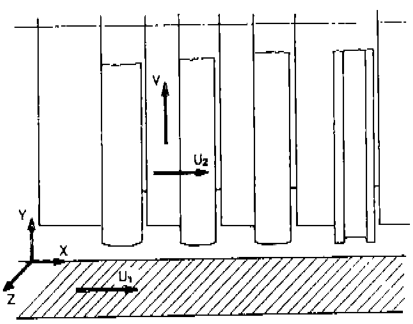

Fig. 1 Surface profile of the lubricating gap

Fig. 1 shows the qualitative description of this clearance by the surface profiles of piston, piston ring and cylinder. It is further described by the actual varying positions of the piston and piston rings in the cylinder during the working cycle. Applying the usual assumptions for sliding bearings for the calculation of the hydrodynamic lubrication the Reynolds differential equation in the general form is valid:

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) = 6\eta(U_1-U_2)\frac{\partial h}{\partial x} + 12\eta V \qquad (1)$$

This partial elliptic differential equation cannot be integrated in a closed form. By using additional simplifying assumptions several analytical solutions [7-19] have been found as already reported. There are existing two groups of calculation models with analytical solutions: On the one hand those which calculate the lubrication conditions exclusively at the piston rings and on the other hand those concerning the unit of piston and piston ring with a joint lubrication film. The lubrication theory as used here and already in an earlier paper [5] concerning the interacting system piston and piston ring includes automatically the special case of having only hydrodynamic lubrication at the piston rings because of lacking oil in the clearance between piston and liner.

Especially for refrigeration compressors often up to certain sizes having only one piston ring the assumption of an oil accumulation before the piston ring can be made and therefore, there exists a joint lubrication film. Visual studies of the clearance gap have proved this assumption. Assuming here a joint lubrication film at piston and piston ring [4] the calculation of lubrication conditions at a trunk piston has to take into account an oil flow in circumferential direction of the piston, because of a different wide clearance at the circumference. Therefore, the second lefthand term of the equation 1 cannot be neglected and the equation must be solved in this case with the help of a non-analytical way of solution.

As already mentioned Burmeister [5] described an iterative numerical procedure of solving the Reynolds equation for the calculation of lubrication conditions at a trunk piston assuming constant piston velocity U and neglecting the second righthand term $12\eta V$ for the squeeze effect. By substituting the differentials of the equation 2

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) = 6\eta U\frac{\partial h}{\partial x} \qquad (2)$$

by their finite difference equivalents the so formed difference equation can be solved by a process with over relaxation. The oil film thickness between piston ring and liner being not constant over the circumference results from the equilibrium of the radial forces acting on the piston ring, Fig. 2.

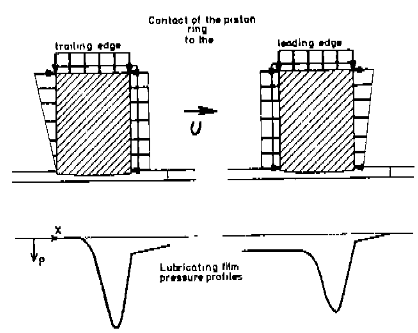

Fig. 2 Pressures acting on the piston ring

The contact of the piston rings in the grooves with the leading or trailing edge depends on the axial equilibrium of the forces acting on the ring which in this first two-dimensional model depends only on the gas forces before and behind the piston ring. Further the spring tension p_E of the piston rings is assumed to be constant over the whole circumference of the piston ring. Because of the only small deformation of the piston ring by different circumferential oil film thicknesses the value of the spring tension corresponding to the nominal diameter is assumed to be valid for the calculations.

SLIDING GEOMETRY

For hydrodynamic pressure build-up in a lubricating gap a converging shape of the clearance is essential. In the case of the system piston-

piston-ring-, cylinder itisformed by different step-shaped or wedge-shaped geometry of the gap, Fig. 1. The known one-dimensional models for calculating only the piston ring lubrication use for the description of the surface profile of the piston rings different circular-, parabolic- or wedge-shaped cross-sections at the piston ring edges. For the one-dimensional calculation with analytical solution for the unit piston-, piston ring-, cylinder with joint lubrication film a wedge-shaped profile of piston and piston ring has been assumed. This assumption is based on surface profile measurements of piston rings after running-in and has also shown a better correlation between calculations with wedge-shaped lubricating film profiles and measurements of ring friction as reported by Horgen [9] . Principally the final effective sliding profile of the piston ring is a result of a wear process during running-in of the engine. Until today there are only a few intentions of the piston ring manufacturers in gaining better sliding conditions of piston rings for lowering the frictional losses by means of a piston ring sliding profile manufactured already with a shape as later formed by wear.

The calculation procedure of Burmeister for the two-dimensional problem allows the numerical input of arbitrary geometries of piston and piston ring profiles into the program because of its independence from analytical treatment. About the influence of the different wedge-shaped ring profiles on the lubrication conditions already has been reported here [ ].

In extension to this statement the calculations of Burmeister for the case of parabolic piston ring edges shows minimal frictional losses at a parabolic exponent of n = 1.25 as shown in Fig. 3, but it can be stated here that for this calculation model no analytical expression of the piston ring shape is necessary because of a numerical input of the film heights so that an arbitrary shape can be used for calculation.

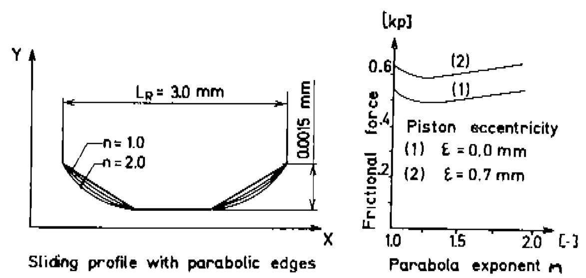

Fig. 3 Piston ring surface with parabolic edges

Concerning the calculation described in this paper as an example for the simulation of the lubrication conditions at the piston of a refrigeration compressor a symmetrical wedge-shaped piston ring profile Fig. (4) is assumed. The wedge-shaped edges of the piston rings correspond to the typical form of profiles as found by Neale [20] and Braendel [21] in measuring a great number of piston ring profiles. The inclination of the wedge has a medium profile angle of about 0.1 degree.

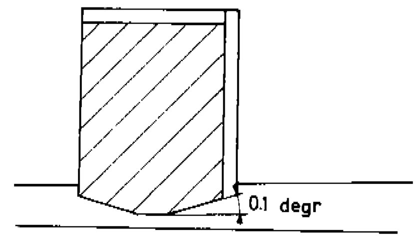

Fig. 4 Symmetrical wedge shaped profile

TOTAL FILLING OF THE PISTON CLEARANCE

To begin with the influence of the pressure gradient in circumferential direction of the piston with resulting oil flow in the z-direction shall be demonstrated. For the calculation following equation (2), the assumption is made that under eccentricity of the piston of ε = 0,7 for this example in the circumferential direction of the piston a constant accumulation of oil before the leading edge of the piston ring is valid. A piston velocity of U = 5 m/s and a dynamic viscosity of η = 98.1 cP were assumed. Fig. 5 shows the plotted diagrams of the three-dimensional pressure profile in the lubrication gap.

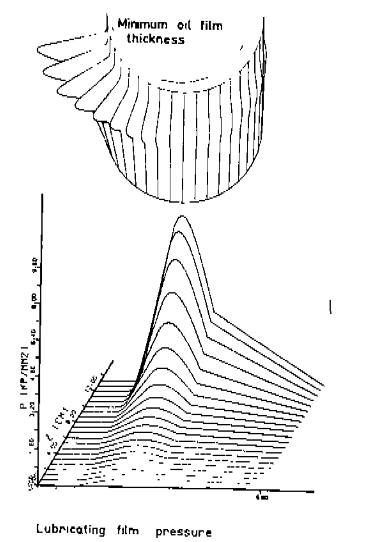

Fig. 5 Pressure profile

In the upper part of the figure the distance piston ring-cylinder is plotted as a significant value for the lubrication conditions in the circumferential direction of the piston. Fig. 6 shows the oil flow in circumferential direction of the piston which is caused by the pressure gradient dp/dz. The main result is that in the region of the piston ring no circumferential oil flow can be stated. Further comparing the results of the one-dimensional and two-dimensional calculation for the totally filled piston clearance assuming a constant oil accumulation only very small deviations of about 1% between both calculations can be stated. This statement is of great importance for the following extension of the calculation model.

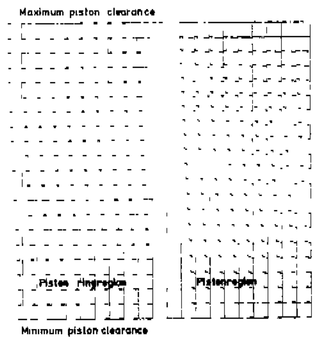

Fig. 6 Oil flow in the lubricating gap

PARTIAL FILLING OF THE PISTON CLEARANCE

The distribution of oil in the lubricating gap has been explored in several visual studies using test engines with glass cylinders. The results of such studies of Jakobs [22] found at a test engine with glass cylinder were evaluated here for the following calculations. These visual studies had been carried out in the laboratories in which the authors are working. The technical data of the experimental rig and the test conditions and results could, therefore, be used completely in contrast to the results of other researches. The assumption of a joint lubrication film at piston and piston ring totally filling the piston clearance in the whole circumferential direction as assumed for the calculation resulting in Fig. 5 is only an idealized hypothesis. Stroboscopic photographs of the piston clearance show a different oil accumulation at the leading edge of the piston ring in circumferential direction of the piston. This different behaviour is especially obvious on the thrust-and antithrust side of the piston and also varies during the piston stroke because of the influences of the longitudinal and transverse piston motion. For the extension of the calculation model in the direction of more realistic conditions the partial filling of the lubrication gap has, therefore, to be concerned and its influence on the lubrication condition has to be studied by calculation. Fig. 7 shows a distribution of oil as stated by Jakobs at a piston with absolute cylindrical geometry as usual in compressors and found at a pressure level corresponding to refrigeration compressors.

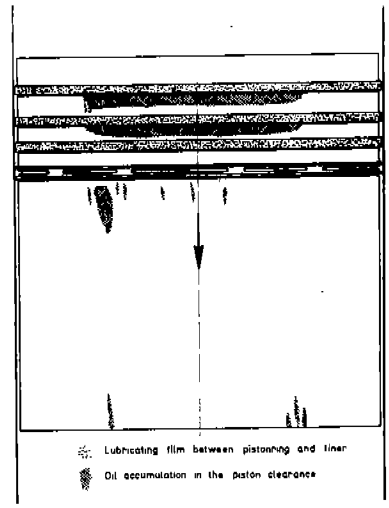

Fig. 7 Oil distribution between piston, piston ring and cylinder

In the region between the piston rings an oil accumulation can be found, which produces hydrodynamic forces being in equilibrium with the thrust forces generated by gas and mass forces of the piston.

In the calculation of the lubrication conditions with totally filled clearance between piston and liner as well as in the case of only being oil between piston ring and cylinder walls for the determination of the equilibrium of forces acting on the piston ring the assumption of constant spring tension p_E is allowed. In the case of only partially filled clearance the elastic deformation of the piston ring must be taken into account. Ting and Mayer [17] treated in their one-dimensional model the piston ring as a thin-walled elastic cylinder. The assumption of such a splitless piston ring can be justified only with considerable lower mathematical difficulties compared with the treating of an actual split ring as commonly used in refrigeration compressors. The transference of this simplifying method to the two-dimensional model cannot be made because of the not given constant circumferential load. Therefore, the piston ring is to be dealt as an elastic circular bowed cantilever.

BENDING THEORY

Using Arnold's theory [23] of self-expanding piston rings the deflection is calculated following the generally known differential equation

$$h''(\varphi) + h(\varphi) = r_m^2 \, (1/\rho - 1/r) \tag{3a}$$

$$h''(\varphi) + h(\varphi) = r_m^2 \, M(\varphi) \, / \, (E \cdot I) \tag{3b}$$

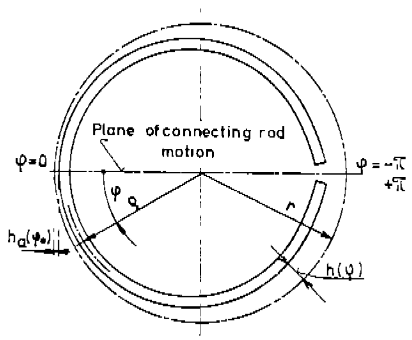

Fig. 8 Piston ring deformatio

The mathematical procedure is indeed considerably complex. The remarkable rise in computing time enforces a restriction of the model before a further extension of the calculation is possible.

Therefore, for the present, the position of the ring gap respectively of the ring back is fixed for calculation into the plane of the connecting rodmotion, Fig. 8. Fig. 9 shows the graphical description of the calculation of the lubrication conditions concerning the first piston ring with an oil accumulation according to Fig. 7.

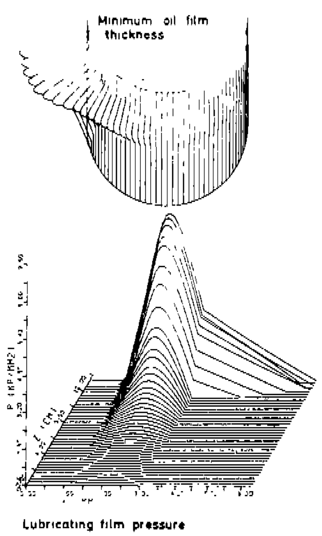

Fig. 9 Pressure profile

THE INFLUENCE OF PISTON TRANSVERSE MOTION

As known from the visual studies of the lubrication conditions at the glass cylinder engine the so-called transverse motion of the piston or secondary motion has an important influence on the distribution of oil in the clearance between piston and liner. Assuming an interacting lubricating film at the piston and piston ring, therefore, a reaction of the piston transverse motion on the lubrication conditions generally and specially on those of the piston ring must be taken into account. The secondary motion of the piston as described in the literature takes place governed by the equilibrium of gas-, mass- and lubricating filmforces. For the optimal layout of the lubrication conditions and the dynamic behaviour of the piston motion with regard to the generation of vibration and noise [26 - 31] the pre-calculation of the piston secondary motion in the equilibrium of all acting forces should be the aim of the calculation of lubrication conditions. At this moment only the influence of the piston secondary motion on the lubrication conditions can be examined because of the less complex calculation program. This motion is composed of translation and rotation at the same time as shown in Fig.10.

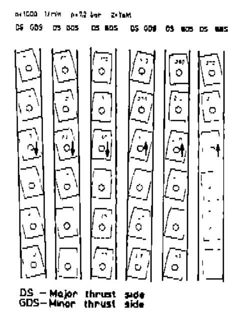

Fig. 10 Piston secondary motion

As input into the calculation for the distance piston liner measured values of Jakobs have been chosen which were recorded under working conditions similar to those in a refrigeration compressor with the refrigerant R 12. They are shown later together with the computed results as gained by the following calculation.

For this calculation of hydrodynamic lubrication under variable speed and load Constantinescu [24] describes the Reynolds differential equation in the following form.

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) + \frac{\partial}{\partial z}\left(h^3\frac{\partial p}{\partial z}\right) = 6\eta\left(2V + U\frac{\partial h}{\partial x} - \frac{\rho}{\eta}\frac{\partial U}{\partial t}\frac{\partial h^3}{\partial x}\right) \quad (4)$$

This equation shows compared to equation 1 an additional acceleration term on the right hand side and also cannot be integrated in a closed form. Deducing from the experiences up to now with the two-dimensional model for unit piston-, piston ring-, cylinder the calculation of the three-dimensional pressure profile as shown in Fig. 5, for a whole working cycle under additional consideration of the transverse velocity V seems to be a too big problem even for big computers. The dimensions of storage capacity and computing time cannot be covered. Therefore, based on the mentioned good correspondence of the one-dimensional and the two-dimensional calculation results the pressure generation circumferential to the piston is substituted by suitable supporting functions. The resulting simplified Reynolds differential equation (5) is also solved

$$\frac{\partial}{\partial x}\left(h^3\frac{\partial p}{\partial x}\right) = 6\eta\left(2V+U\frac{\partial h}{\partial x} - \frac{\rho}{\eta}\,\frac{\partial U}{\partial t}\,\frac{\partial h^3}{\partial x}\right) \qquad (5)$$

by the finite difference method and calculation of the lubrication film pressure by the process with over-relaxation. For the major and minor thrustside the pressure graphs are calculated and the whole pressure profile is found out by interpolation with the help of the above mentioned supporting functions. The variable load of the sliding system piston liner results from the varying gas-and mass forces acting on the piston during the working cycle. The cylinder pressure of already built engines can be indicated by measurements. In the state of design of a refrigeration compressor it is possible to predetermine this pressure by means of a math matical model for compressor simulation leading to a computed p, V - diagram as shown in Fig. 11.

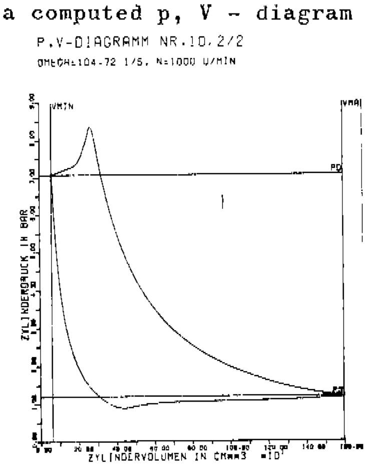

Fig. 11 Simulated p,v - diagram

The condensation and evaporation temperature of the simulated working process and the geometric dimensions of a R 12 refrigeration compressor have been chosen so that the calculated pressure are equal in their heights to those of the glass cylinder engine because of possible correspondence of experimental results from that engine to the compressor model. The force acting perpendicularly from the piston to the cylinder wall respectively the bearing lubricating film is given by the following equations according to Fig. 12.

$$F_{GAS}=P_{GAS}\cdot A_K \qquad (6)$$

$$F_{AX} = -m_{os}\,\ddot{x} + F_{GAS} \qquad (7)$$

$$F_N = F_{AX}\cdot \tan\psi \qquad (8)$$

$$x(\varphi)=r(1-\cos\varphi+\frac{\lambda}{2}\sin^2\varphi) \qquad (9)$$

$$\dot{x}(\varphi)=r\omega(\sin\varphi+\frac{\lambda}{2}\sin 2\varphi) \qquad (10)$$

$$\ddot{x}(\varphi)=r\omega^2(\cos\varphi+\lambda\cos 2\varphi) \qquad (11)$$

$$\sin\psi=\lambda\sin\varphi \qquad (12)$$

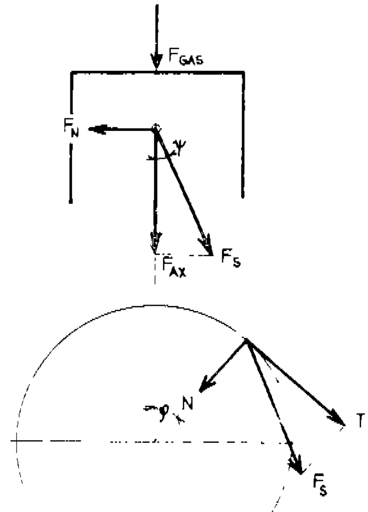

Fig. 12 Cinematics of crank mechanism

EQUILIBRIUM OF FORCES AND MOMENTS ACTING ON THE PISTON

The equilibrium of forces acting in the plane of the connecting rod motion, in which the piston secondary motion is supposed to take place is given by equations

$$\sum F_{pi} = F_N - m\ddot{y} \qquad (13)$$

$$F_{pi} = \int p_i \cdot dA\cdot\cos\alpha_i \qquad (14)$$

according to Fig. 13.

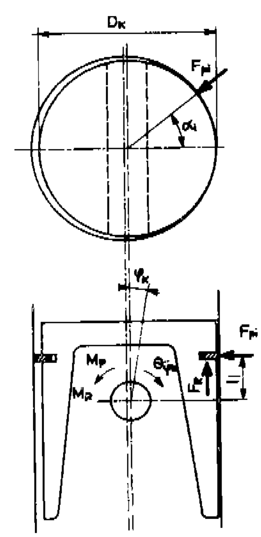

Fig. 13 Forces and moments acting on the piston

The equilibrium of moments referred to the gudgeon pin axis becomes

$$M_P + M_R = \Theta_K\,\ddot{\varphi}_K \qquad (15)$$

in which the bearing capacity of the lubricating film exerts the moment

$$M_P = \sum F_{pi}\cdot l_i \qquad (16)$$

and the frictional moment is

$$M_R = M_B + M_{SF} \tag{17}$$

The sharing part of the friction torque at the gudgeon pin

$$M_B = (m_{os}\ddot{x} - F_{GAS})\,\mu\, r_B \,/\sqrt{1 - \lambda^2 \sin^2\varphi} \tag{18}$$

results from the statement according to Coulombs law of friction with the radius of the gudgeon pin r_B and the coefficient of friction μ.

The other component produced by the lubricating film yields

$$M_{SF} = \sum [\int \tau_x \cdot dA \cdot D_K/2 \ \cos\alpha_i] \tag{19}$$

The following assumptions are made here:

a) The gas force F_{Gas} is acting to the centre of the piston and causes no moment to the gudgeon pin

b) There is no gudgeon pin offset

c) The centre of gravity of the masses accelerated in the y - direction lies in the height of the gudgeon pin axis

The bearing capacity of the lubricating film between piston, piston ring and cylinder is computed by integration of the pressure profile, Fig. 5, and by summation of the normal to the gudgeon pin axis acting cosine terms of the forces, see equation (13).

EQUILIBRIUM OF FORCES ACTING ON THE PISTON RING

During the reciprocating motion of the piston the piston rings are subjected to gas-, mass- and lubricating film-forces of alternating directions. For that reason motions of the piston rings relative to the piston occur with simultaneous change of the axial contact of the piston ring in the groove. These relative motions mainly take place in the near of the dead centres and can be verified by the following equation

$$F_{Pax} - F_{Tr} - F_W + F_R = m_r\, a_r \tag{20}$$

F_{Pax} resulting axial gas force

F_{Tr} inertia of the piston ring

F_W resulting axial contact force

F_R frictional force acting from the cylinder to the piston ring

m_r mass of the piston ring

a_r relative acceleration of the piston ring

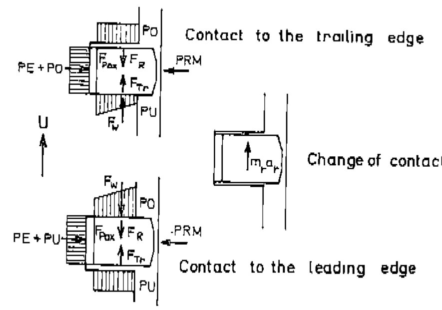

Fig. 14 Forces acting on the piston ring

Fig. 14 shows the forces acting on the piston ring and the change of the axial contact of the ring when the relative acceleration of the piston ring changes its direction according to equation

$$a_r = \frac{F_{Pax} - F_{Tr} - F_N + F_R}{m_r} \tag{20}$$

Consequently the radial equilibrium of forces is dependent on changing conditions with regard to the efficient gas pressure.

CALCULATION OF THE MINIMUM OIL FILM THICKNESS BETWEEN PISTON RING AND LINER

Here, as an example for the application of the model to a special case the lubrication conditions at the piston of the glass cylinder engine of Jakobs shall be simulated mathematically for running conditions with a two stroke working cycle and with minimum and maximum pressures corresponding to the refrigeration compressor conditions. For the calculation of the lubrication conditions the graphs of the cylinder pressure and the piston transverse motion, as described in the Figs. 15, 16, 17 serve as input data.

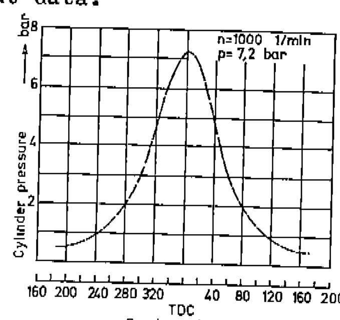

Fig. 15 Gas pressure in the glass cylinder engine

The calculation following equation (5) has been carried out fora whole working cycle with varying gas pressure and with the measured secondary motion of the piston. In a step by step process the position of the piston in the cylinder is prescribed and the minimum oil film thickness between piston ring and liner is calculated iteratively. Hereby the distance piston ring-cylinder must be varied. The results concerning the minimum oil film thickness of the first piston ring with its most severe lubrication conditions are shown in Fig. 16 and 17 for the major and the minor thrust side.

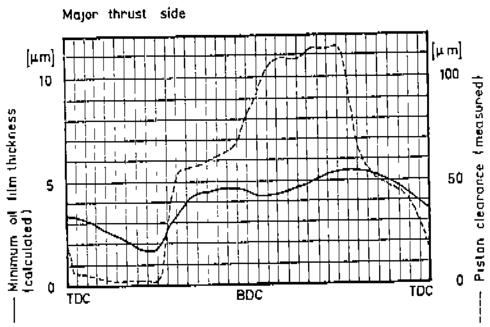

Fig. 16 Minimum oil film thickness maj. thrusts.

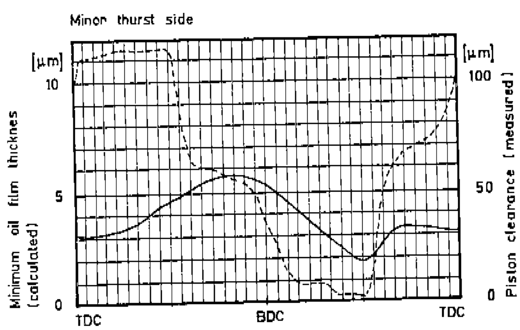

Fig. 17 Minimum oil film thickness min. thrusts.

They are plotted as a function of the crank angle together with the measured piston transverse motion used as input value for calculation. This motion given by opposite displacements on the major and minor thrustside is characterized by the dotted lines and can be demonstrated by the piston positions in Fig. 10.

Qualitatively the minimum oil film thickness at the piston ring shows a similar graph. The maximum values of about 5 μm well correspond with the measurements of Hamilton and Moore [25]. The minimum values of about 2 μm on both sides occur when the piston is in a position very close to the liner at the same side.

The qualitative differences of the graph for piston and piston ring with certain time delays can be explained by the squeeze effect caused by piston transverse motion. Its computation well correlates with the measured values given by Hamilton and Moore and is smaller by a factor of about 10^4 compared with the wedge effect caused by longitudinal piston velocity.

VERIFICATION OF THE CALCULATION MODEL

As mentioned above the final aim of the simulation model described in this paper is to predetermine the lubrication conditions of trunk pistons in refrigeration compressors. A suitable method has to include the influence of piston secondary motion or even better the predetermination of this motion in the equilibrium of forces and moments acting on the piston assembly.

In the actual state of development the mathematical model must be verified regarding the correlation of computed results to experimental research data. Concerning refrigeration compressors there are no known experimental results of investigation as, for example, measured minimum oil film thickness or other significant values.

Particular details of the oil distribution in the clearance piston-cylinder are lacking for compressors.

Owing to these reasons the verification for the calculation of lubrication conditions was carried out in comparing the calculated load capacity of the lubrication film in the already mentioned glass cylinder engine to the acting forces as described by equations (13) and (14). The result is represented by Fig. 18.

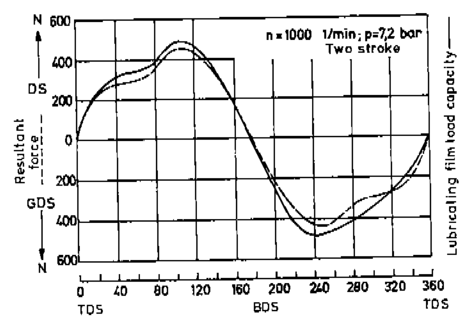

Fig. 18 Comparison of forces

and shows a quite good correlation of lubricating film load capacity with the resultant of gas- and mass forces in the direction of the cylinder wall.

The further extension of the above described simulation of lubrication conditions with special view to refrigeration compressors will be prepared in that way, that basing on the assumption of optimal oil distribution in the clearance piston-cylinder and on a simulated working cycle the piston secondary motion and lubrication conditions can be predetermined by iterative computation of the equilibrium of gas-, mass and lubricating film forces over a whole working cycle.

In this computing program in a step by step process beginning with a supposed position of the piston in the cylinder at a starting point near the middle of the stroke for each incremental crank angle the clearance piston-cylinder and the relative angle of axis have to be pointed precisely for gaining the position of equilibrium.

CONCLUSION

The simulation of lubrication conditions at trunk pistons of refrigeration compressors is very important with regard to compressor optimization, reliability and application limits. In this paper the influences of partially filled lubrication gap and the effect of piston secondary motion are described. At the present state it is already possible to calculate the lubrication conditions over a whole working cycle if the gas pressure, piston velocity, oil distribution in the clearance piston-cylinder and the actual position of the piston in the cylinder are known as a function of the stroke. A way is shown for predetermination of optimized lubrication conditions in the phase of machine design based on simulation of the working cycle of refrigeration compressors.

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